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Description

The present invention relates to a belt type continuous reduction gear mechanism in which a V-belt is made to run under tension between such drive and driven V-pulleys as are each equipped with a fixed pulley half and a movable pulley half made movable back and forth relative to the fixed half for adjustment of an effective radius.

In the continuous reduction gear mechanism of the aforementioned type according to the prior art, there has been widely adopted a one-side following system in which only the drive V-pulley is equipped with an actuator for moving the movable pulley half back and forth so that the forward and backward movements of the movable pulley half at the drive side may be transmitted during reduction through the V-belt to the movable pulley half at the driven side thereby to make the latter back and forth. The one-side following system experiences disadvantageous phenomena such as deformation, heat generation or wear, because its V-belt is subjected to an excessive tension, so that the V-belt has not only its transmission efficiency remarkably deteriorated but also its lifetime shortened, and the system has another defect in that movable pulley half at the driven side has its following operation delayed to degrade the running performance of a vehicle provided with the system.

In DE—A—2 944 928 as defined in the first part of claim 1 there is disclosed a belt type continuous reduction gear mechanism comprising a drive V-pulley arranged coaxially with a start clutch which is arranged coaxially with the crankshaft of an engine and is constructed as a disc clutch; a driven V-pulley connected to an auxiliary gear mechanism of plural-step type; a V-belt made to run under tension between the two V-pulleys; a casing accommodating at least the engine, the start clutch and both drive and driven V-pulleys each of which is equipped with a movable pulley half that can be moved back and forth relative to a fixed pulley half; a drive pulley shaft arranged coaxially with the crankshaft and on which are mounted both the fixed pulley half of the drive V-pulley at the inner side and the movable pulley half of the drive V-pulley at the outer side, this movable pulley half carrying an actuator for adjusting the effective radius and this fixed pulley half being connected to the start clutch; and a driven pulley shaft arranged in parallel with the drive pulley shaft arranged in parallel with the drive pulley shaft and on which are mounted both the fixed pulley half of the driven V-pulley at the outer side and the movable pulley half of the driven V-pulley at the inner side, this movable pulley half carrying an actuator for adjusting the effective radius.

According to the present invention, as defined in claim 1, there is provided a belt type continuous reduction gear mechanism comprising a drive V-pulley arranged coaxially with a start clutch which is arranged coaxially with the crankshaft of an engine and is construct d as a disc clutch; a driven V-pulley connected to an auxiliary gear mechanism of plural-step type; a V-belt made to run under tension between the two V-pulleys; a casing accommodating at least the engine, the start clutch and both drive and driven V-pulleys each of which is equipped with a movable pulley half that can be moved back and forth relative to a fixed pulley half; a drive pulley shaft arranged coaxially with the crankshaft and on which are mounted both the fixed pulley half of the drive V-pulley at the inner side and the movable pulley half of the drive V-pulley at the outer side, this movable pulley half carrying an actuator for adjusting the effective radius and this fixed pulley half being connected to the start clutch; and a driven pulley shaft arranged in parallel with the drive pulley shaft and on which are mounted both the fixed pulley half of the driven V-pulley at the outer side and the movable pulley half of the driven V-pulley at the inner side, this movable pulley half carrying an actuator for adjusting the effective radius; characterised in that the drive V-pulley has its external diameter reduced to substantially the same as that of the start clutch thereby to set the reduction ratio range at a relatively small value; and in that the auxiliary gear mechanism is of plural step reduction type for compensating the shortage of said reduction ratio range; wherein the start clutch is of hydraulic multi-disc type; wherein an oil pump is connected through a control line to the start clutch; and wherein there is disposed in the control line a clutch valve for controlling the discharge pressure of the oil pump so that the connection oil pressure of the starch clutch is varied proportionately of the squeezed extent of a clutch lever.

By making the movable halves of both the drive and driven V-pulleys simultaneously movable back and forth through respective actuators there is provided an effective continuous reduction gear mechanism in which it is possible to improve the transmission efficiency and lifetime of the V-belt, to eliminate the delay in the reduction and to improve the vehicular running performance; it is possible to prevent the V-belt from being slackened (thereby to have its lifetime elongated) by applying a pre-tension to the V-belt when the actuators are not operated yet; it is possible to improve the durabilities of bearings for bearing either the actuators or pulley shafts on a casing; it is possible to elongate the lifetime of the V-belt and to make the power transmission efficient by ensuring the continuous reduction, and with a control system that is compact; it is possible to reduce the size of the drive V-pulley by setting the reduction range of the drive V-pulley at a relatively small value thereby to reduce the external diameter of this pulley and to retain the necessary reduction ratio by compensating the shortage in the reduction range of the drive V-pulley by means of an auxiliary reduction gear mechanism; it is possible to shorten the inter-bearing sizes of the drive and driven V-pulleys and to make their maintenance efficient; it is possible to reduce the

external size of a casing accommodating the drive and driven V-pulleys; it is possible to reduce the size of the start clutch, to which the continuous reduction gear mechanism is to be connected, and to reduce the operating load thereupon thereby to smoothen and maintain the operation of the start clutch in a normal state at all times because this clutch is a hydraulic type multi-disc clutch; and it is possible to improve the durability of a passage through which working oil is to be supplied to the start clutch.

For a better understanding of the invention and to show how its same may be carried into effect, reference will now be made, by way of example, to the accompanying drawings, in which:

Figure 1 is a schematic top plan view showing a power transmission system of a motorcycle;

Figure 2 is a partially and longitudinally sectional top plan view showing a power unit in the power transmission system; and

Figure 3 is an enlarged longitudinally sectional top plan view showing a V-belt type continuous reduction gear mechanism in the power unit.

The present invention will be described in the following in connection with one embodiment thereof, in which it is applied to a motorcycle, with reference to the accompanying drawings. With first reference to Fig. 1, the power unit Pu of the motorcycle is constructed of: an engine E; a start clutch Sc of hydraulic multi-disc type connected outside of the engine E to the crankshaft 1 of the same; a hydraulic V-belt type continuous reduction gear mechanism Tm disposed outside of and connected to the start clutch Sc; and an auxiliary reduction gear mechanism Ta having a plurality of or two (in the shown embodiment) gear trains and disposed in and connected to the continuous reduction gear mechanism Tm, all of which are accommodated in a casing C supported on the not-shown body of the motorcycle.

The casing C is divided, as shown in Fig. 2, into a main case C_1 accommodating both the crank portion of the crankshaft 1 of the engine E and the auxiliary reduction gear mechanism Ta, an auxiliary case C_2 abutting against the outer side of the main case C_1 and accommodating both the continuous reduction gear mechanism T_m equipped with drive and driven V-pulleys 40 and 41 and a V-belt 42 and the start clutch S_c, and a cover C_3 closing the outer side of that auxiliary case C_2 and accommodating a coaction mechanism 130 for the drive and driven V-pulleys 40 and 41. Moreover, the crankshaft 1 and other various rotary shafts of the power unit Pu are all arranged in parallel with the axis of a rear wheel Wr, which is rotatably suspended from the not-shown motorcycle body at the back of the power unit Pu, such that the output shaft of the power unit Pu, i.e., the output shaft 141 of the auxiliary reduction gear mechanism Ta drives the rear wheel Wr through a chain transmission mechanism M.

In order to supply the start clutch Sc and the continuous reduction gear mechanism Tm with working oil, a control line Lc leading from a clutch valv Vc is conn cted with the start clutch Sc, and first and second lines L_1 and L_2 leading from an oil pump P acting as an oil pressure source are connected with the drive and driven portions of the continuous reduction gear mechanism Tm, respectively.

In Fig. 1, incid ntally, reference letters Vr indicate a relief valve for the oil pump P, and letter R indicates an oil sump which is formed in the bottom of the casing C.

The constructions of the respective portions of the power unit Pu will be consecutively described with reference to Figs. 2 and 3.

First of all, the start clutch Sc will be described in the following. This start clutch Sc is mounted on the crankshaft 1 adjacent to the outer side of the most righthand bearing 2 which bears the crankshaft 1 functioning as a rotary shaft. That start clutch Sc is constructed of: an outer clutch member 4, which is splined at 3 to the crankshaft 1, and an inner clutch member 5 which is made integral with the fixed pulley half 44 of the drive V-pulley 40, which will be described hereinafter. Between those outer and inner clutch members 4 and 5, there are interposed a plurality of drive friction discs 6, which are slidably splined to the outer clutch member 4, and a plurality of driven friction discs 7, which are slidably splined to the inner clutch member 5, such that the friction discs 6 and 7 are alternately positioned in a coextensive manner. On the outer clutch member 4, there is retained a pressure-receiving ring 8 for restricting the outermost drive friction disc 6 from moving to the outside. The outer clutch member 4 is formed at the opposite side to that pressure-receiving ring 8 with a hydraulic cylinder 9, in which there is slidably fitted such a piston 11 as faces the innermost drive friction disc 6 through a dish-shaped buffer spring 10. That piston 11 is biased in the backward direction, i.e., apart from the groups of the friction discs 6 and 7 by the action of a return spring 12 which is disposed inside of the inner clutch member 5. That hydraulic cylinder 9 has its pressure oil chamber 13 supplied with the working oil from the aforementioned control line Lc by way of a working oil passage 14 which is formed in the axial end portion of the crankshaft 1.

Thus, the hydraulic cylinder 9 and the piston 11 constitute together a hydraulic actuator mechanism. When the oil pressure chamber 13 is supplied with the working oil under a high pressure, the piston 11 receives that oil pressure to advance, while compressing the return spring 12, so that it thrusts the grouped drive and driven friction discs 6 and 7 toward the pressure-receiving ring 8 thereby to frictionally couple the adjoining friction discs 6 and 7 through a partially clutch-applied state. Under this clutch-applied state, the power transmitted from the crankshaft 1 to the outer clutch member 4 is further transmitted through the two groups of the friction discs 6 and 7 to the inner clutch member 5 and further to the continuous reduction gear mechanism Tm at the subsequent step. If the working oil is released from the hydraulic cylinder 9, on the other hand

the piston 11 is retracted by the elastic force of the return spring 12 so that the frictional c uplings between the friction discs 6 and 7 are released (to a clutch-blocked state) to interrupt the aforementioned power transmission.

The start clutch Sc adopts the wet type in which the friction discs 6 and 7 are cooled down with the working oil. Now, if the cooling oil to be supplied to the friction discs 6 and 7 is excessive, trailing phenomena are caused, when the clutch is blocked, between the friction discs 6 and 7 by the viscosity of the cooling oil whereas slippage is liable to take place, when the clutch is applied, between the friction discs 6 and 7. If the cooling oil is short, on the contrary, the respective friction discs 6 and 7 are undesirably overheated during the partially applied operation in which much frictional heat is generated. Therefore, the flow rate of the cooling oil to be supplied is required to be controlled at a zero or low level, when the clutch is blocked or connected, and at a high level when the clutch is partially applied. For this control, there is provided a flow regulating valve 15.

This flow regulating valve 15 is made to have a cylindrical shape and is slidably fitted in the aforementioned working oil passage 14 of the crankshaft 1 so that said valve 15 has its lefthand end face subjected to the oil pressure in the working oil passage 14 and its righthand end face subjected to the atmospheric pressure and the elastic force of a return spring 16. That flow regulating valve 15 is formed with a valve port 17 which has communication with the working oil passage 14. The crankshaft 1 is formed with a first oil port 18 with an orifice 18, which has communication with the aforementioned valve port 17 when said valve 15 is moved to a predetermined righthand position. Moreover, the outer clutch member 4 is formed with a second oil port 20 which always effects communication of the first oil port 19 with the inside of the inner clutch member 5 through the spline 3. The orificed first oil port 19, the spline 3 and the second oil port 20 form together a cooling oil passage which is branched from the working oil passage 14.

Thus, during the clutch-blocked operation in which the working oil passage 14 is under a low pressure, the regulating valve 15 is retained at its lefthand limit by the force of the return spring 16 so that the communication between the valve port 17 and the first oil port 19 is either blocked, as shown, or suitably throttled, whereby the flow rate of the cooling oil to be supplied from the working oil passage 14 to the start clutch Sc is regulated to a zero or low level. When the oil pressure in the working oil passage 14 is boosted to such a level as to invite the partially clutch-applied state, the regulating valve 15 is moved rightwardly, while compressing the return spring 16, by the boosted oil pressure to establish communication of the valve port 17 with the first oil port 19, whereby the cooling oil is sufficiently supplied from the working oil passage 14 to the start clutch Sc by way of the valve port 17 and the first and second oil ports 19 and 20. The maximum flow rate of the cooling oil at this time is regulated by the action of the orifice 18. If the oil pressure in the working oil passage 14 is further boosted to invite the clutch-applied state so that the regulating valve 15 is further moved to the right, the communication between the valve port 17 and the first oil port 19 is again either blocked or suitably throttled, whereby the flow rate of the cooling oil is again regulated to zero or low level.

Reverting to Figure 1, the description will be made upon the clutch valve Vc for operating the aforementioned start clutch Sc. Into a cylindrical valve housing 25 having its one end closed, there are consecutively inserted a return spring 26, a valve spool 27, a pressure regulating valve 28 and a push plate 29. To this push plate 29 at the outermost position, there is connected one end of an actuating lever 31 which has its central portion supported by a fixed pivot pin 30. To the other end of the actuating lever 31, there are connected both a control wire 33, which leads to a clutch lever 32 attached to a steering handle H, and an actuating spring 34. This actuating spring 34 has a stronger force than the aforementioned pressure regulating spring 28 so that it can push the pressure regulating spring 28 through the actuating lever 31 and the push plate 29 in accordance with the release of the clutch lever 32 thereby to increase the set load upon the spring 28.

The valve housing 25 is formed with first to fourth ports 35_1 to 35_4 which are opened in the inner wall thereof such that they are arranged side by side from the pressure regulating spring 28. The first port 35_1 has communication with the oil sump R; the second port 35_2 provides a lead to the control line Lc; the third port 35_3 has communication with the oil pump P; and the fourth port 35_4 has communications with not only the control line Lc through an orifice 36 but also with a reaction oil pressure chamber 38 which is formed in the valve housing 25 for accommodating the return spring 26. On the other hand, the valve spool 27 is formed with an annular groove 39 which is sized and positioned to switch the communication of the aforementioned second port 35_2 with the first or third port 35_1 or 35_3.

Thus, if the actuating lever 31 is sufficiently retracted from the push plate 29 against the force of the actuating spring 34, as shown in Figure 1, by pulling the clutch lever 32 toward the steering handle H, the valve spool 27 is moved to the right by the return spring 26 to close the third port 35_3 but to establish the communication between the first and second ports 35_1 and 35_2. As a result, the pressure in the hydraulic cylinder 9 of the start clutch Sc is released into the oil sump R so that the start clutch Sc comes into its blocked state.

When the operating force of the clutch lever 32 is gradually released so that the push plate 29 is forced to push the pressure regulating valve 28 by the actuating spring 34, the valve spool 27 is moved to the left to close the first port 35_1 but to establish the communication between th second and third ports 35_2 and 35_3 so that the oil dis-

charged from the oil pump P is introduced into the control line Lc. If the oil pressure of the control line Lc is accordingly boosted, this boosted pressure is introduced through the orifice 36 into the reaction oil pressure chamber 38 so that the valve spool 27 is forced to return to the right to a position in which the pushing force of that oil pressure is balanced with the set load of the pressure regulating spring 28. Therefore, the oil pressure of the control line Lc, i.e., the applying oil pressure of the start clutch Sc can be boosted in accordance with such increase in the set load of the pressure regulating spring 28 as accords with the returning motion of the clutch lever 32.

If there is used in the aforementioned manner the clutch valve Vc which can vary the applying oil pressure for the start clutch Sc proportionately of the squeezed extent of the clutch lever 32, the oil pressure for applying the start clutch Sc can be set at a sufficiently high level substantially independently of the force for operating the clutch lever 32, even if that force is lightly set, whereby the size of the start clutch Sc can be reduced. This size reduction is further promoted by mounting the start clutch Sc on the crankshaft 1 which has the highest r.p.m. and the lowest torque of the power unit Pu, as has been described hereinbefore. Moreover, since the applying oil pressure of the start clutch Sc after the application is proportional to the r.p.m. of the engine E, it is possible to attain the clutch load matching with the engine torque thereby to improve the durability of the clutch system.

The continuous reduction gear mechanism Tm will be described in the following. This continuous reduction gear mechanism is composed mainly of: the drive V-pulley 40 which is mounted on the crankshaft 1 adjacent to the righthand side of the start clutch Sc; the driven V-pulley 41 which is arranged at the back of and adjacent to the drive V-pulley 40; and the V-belt 42 which is made to run between the two V-pulleys 40 and 41.

The drive V-pulley 40 has its external diameter so reduced to set its reduction ratio range at a relatively small value as to be substantially equal to that of the outer clutch member 4 of the start clutch Sc. The drive V-pulley 40 is composed of: the fixed pulley half 44 which is rotatably borne on the righthand end portion of the crankshaft 1 by means of a bearing 43; and the movable pulley half 47 which is slidably connected through two ball keys 46 to a hollow drive pulley shaft 45 made integral with that fixed pulley half 44 and disposed coaxially with the crankshaft 1. On the rear portion of that movable pulley half 47, there is fixed by means of screws 48 a piston 49. A hydraulic cylinder 50 accommodating that piston 49 has its rear wall plate 50a borne in the auxiliary case C_2 by means of a ball bearing 51 and connected to the drive pulley 45 by means of a retainer ring 52 acting as a pressure-receiving member. The piston 49 partitions the inside of the hydraulic cylinder 50 into a first oil pressure chamber 50_1 at the side of the fixed pulley half 44 and a second oil pressure chamber 50_2 at the opposite side. The pressure-receiving face of the piston 49 is formed to be wider at the second oil pressure chamber 50_2 than at the first oil pressure chamber 50_1.

Therefore, if an equal oil pressure is introduced into the two oil pressure chambers 50_1 and 50_2, the piston 49 is moved to the left by the oil pressure difference due to the difference between the right and left pressure-receiving areas, whereby the movable pulley half 47 is moved forward toward the fixed pulley half 44 so that the effective radius of the drive V-pulley 40, i.e., the contact area with the V-belt 42 can be enlarged. On the other hand, if the oil pressure of the second oil pressure chamber 50_2 is released in a state in which the oil pressure is applied to the first oil pressure chamber 50_1, the piston 49 is moved to the right by the oil pressure in the first oil pressure chamber 50_1, whereby the movable pulley half 47 is retracted apart from the fixed pulley half 44 so that the effective radius of the drive V-pulley 40 can be reduced. Thus, the piston 49 and the hydraulic cylinder 50 constitute together a hydraulic actuator Hyl for moving the movable pulley half 47 back and forth relative to the fixed pulley half 44. For the hydraulic operation of the piston 49 thus far described, there is disposed in the drive pulley shaft 45 a first control valve V_1, the detail of which will be described hereinafter.

The hydraulic cylinder 50 is placed in an integral relationship with the fixed pulley half 44, too, because its rear wall plate 50a is connected to the drive pulley shaft 45 by means of the retainer ring 52, as has been described hereinbefore. With this relationship, the thrust load, which is to be exerted between the fixed pulley half 44 and the hydraulic cylinder 50 in accordance with the hydraulic operation of the piston 49, can be transmitted to and borne on the drive pulley shaft 45. As a result, the load upon the ball bearing 51 for rotatably bearing the hydraulic cylinder 50 is lightened so that the durability of the same can be improved.

On the other hand, the driven V-pulley 41 is composed of: a fixed pulley half 57, which is made integral with a hollow driven pulley shaft 56 arranged in parallel with the drive pulley shaft 45, and a movable pulley half 59 which is axially slidably connected to the follower pulley shaft 56 by means of three ball keys 58. Moreover, the fixed pulley half 57 is arranged at the back of the externally positioned movable pulley half 47 of the drive V-pulley 40, and the movable pulley half 59 is arranged at the back of the internally positioned fixed pulley half 44 of the same. To the back of the movable pulley half 59, there is fixed by means of screws 60 a piston 61. A hydraulic cylinder 62 accommodating that piston 61 is connected to the driven pulley shaft 56 by means of a retainer ring 63 acting as a pressure-receiving member. The piston 61 partitions the inside of the hydraulic cylinder 62 into a first oil pressure chamber 62_1 at the side of the fixed pulley half 57 and a second oil pressure chamber 62_2 at the

opposite side. The pressure-receiving face of the piston 61 is formed to be wider at the second oil pressure chamber 62_1 than at the first oil pressure chamber 62_1.

Therefore, if an equal pressure is introduced into the two oil pressure chambers 62_1 and 62_2, the piston 61 is moved to the right by the oil pressure difference due to the difference between the right and left pressure-receiving areas, whereby the movable pulley half 59 is moved forward to approach the fixed puley half 57 so that the effective radius of the driven V-pulley 41 can be enlarged. On the other hand, if the oil pressure of the second oil pressure chamber 62_2 is released in a state in which the oil pressure is applied to the first oil pressure chamber 62_1, the piston 61 is moved to the left by the oil pressure of the first oil pressure chamber 62_1, whereby the movable pulley half 59 is retracted apart from the fixed pulley half 57 so that the effective radius of the driven V-pulley 41 can be reduced. Thus, the piston 61 and the hydraulic cylinder 62 constitute together a hydraulic actuator Hy2 for moving the movable pulley half 59 back and forth relative to the fixed pulley half 57. For the hydraulic operation of the piston 61 thus far described, there is disposed in the driven pulley shaft 56 a second control valve V_2, the detail of which will be described hereinafter.

The driven pulley shaft 56 is borne in the casing C at its righthand and lefthand end portions and at its central portion by means of bearings 64, 65 and 66. Between the central bearing 65 and the righthand bearing 66, moreover, the hydraulic cylinder 62 is placed in an integral relationship with the fixed pulley half 57 by means of the retainer ring 63 and the driven pulley shaft 56. With this relationship, the thrust load, which is to be exerted between the fixed pulley half 57 and the hydraulic cylinder 62 in accordance with the hydraulic operation of the piston 61, can be transmitted to and borne by the driven pulleyshaft 56. As a result, the load upon the bearings 65 and 66 can be lightened so that the durability of the same can be improved.

On the other hand, if the continuous reduction gear mechanism Tm is accommodated in the auxiliary case C_2, its inter-bearing size can be limited within the width of the auxiliary case C_2 to shorten the drive and driven pulley shafts 45 and 56 and to improve the durabilities of the same. Moreover, since the coaction mechanism 130 having a relatively high frequency for its maintenance is disposed outside of the casing C whereas the continuous reduction gear mechanism Tm having a next frequency is disposed at an intermediate position, their maintenances can be efficiently conducted.

Now, the description is changed to the first and second control valves V_1 and V_2 and to their surrounding oil passages.

The first control valve V_1 is composed of: a cylindrical follower valve spool 71, which is slidably fitted in the hollow drive pulley shaft 45; and a cylindrical main valve spool 70 which is slidably fitted in that follower valve spool 71. In the main valve spool 70, there are inserted inner and outer communication conduits 72 and 73 which are fitted the former in the latter. The inner communication conduit 72 extends to the right and left through the main valve spool 70 and has its lefthand end fitted in the crankshaft 1 through the flow regulating valve 15 to communicate with the working oil passage 14 and its righthand end connected to the aforementioned control line Lc which is formed in the cover C_3 of the casing C.

Moreover, the inner communication conduit 72 defines inside of the main valve spool 70 a cylindrical oil passage 74, which has communication with the aforementioned first oil line L_1 formed in the cover C_3 by way of the outer communication conduit 73.

Those two communication conduits 72 and 73 are connected partly by reducing the righthand end of the outer one 73 and by welding the righthand outer circumference of the inner one 72. Moreover, a mounting flange 75 is welded to the outer circumference of the outer communication conduit 73. The cover C_3 has its inner wall formed with a stepped oil port 76 of the first oil line L_1, which is aligned with the axis of the drive pulley shaft 45. The mounting flange 75 is fitted in the larger portion of the stepped oil port 76 by means of an O-ring 77 acting as an elastic seal member and is prevented from coming out by means of a retainer ring 78. On the other hand, the rightward protrusion of the inner communication conduit 72 is fitted in the reduced portion of the stepped oil port 76 by means of an O-ring 79 acting as an elastic seal member. Thus, the dual communication conduits 72 and 73 are supported in the cover C_3 in such a floating manner that they can follow the fluctuations of the center axes of the crankshaft 1 and the drive pulley shaft 45 thereby to prevent themselves, and the crankshaft 1 and the drive pulley shaft 45 from being worn due to the abutments inbetween. Incidentally, reference numeral 80 indicates a communication hole which is formed in the circumferential wall of said communication conduit 73 so as to establish communication between the stepped oil port 76 and the inside of the outer communication conduit 73.

The main valve spool 70 has its outer circumference formed with a pair of left and right annular oil supply grooves 81 and 82 and an annular oil release groove 83. The oilk supply grooves 81 and 82 are made to communicate with the cylindrical oil passage 74 in the main valve spool 70 by way of holes 84 and 85. On the other hand, the follower valve spool 71 has its outer circumference formed with a pair of left and right annular oil grooves 86 and 87. The left oil groove 86 is made to always communicate not only with left oil supply groove 81 of the main valve spool 70 by way of a hole 88 but also with the first oil pressure chamber 50_1 of the hydraulic cylinder 50 by way of a hole 89, an annular oil passage 90 and an oil passage 91. On the oth r hand, the right oil groove 87 is made to always communicate not

only with th oil release groove 83 of the main valve spool 70 by way of a hole 92 but also with the second oil pressure chamber 50_2 of the hydraulic cylinder 50 by way of a hole 93. Moreover, the follower valve spool 71 is formed with a hole 94 for the oil release system, which is made operative to provide and block the communication between the right oil groove 87 and the right oil supply groove 82 of the main valve spool 70, and a notched oil release port 95 for providing and blocking the communication between the oil release groove 83 of the main valve spool 70 and the inside of the casing C. Furthermore, the follower valve spool 71 is connected to the movable pulley half 47 by means of a coaction pin 96 which extends radially of the drive pulley shaft 45 so that it can move to the right and left together with the latter. That portion of the drive pulley shaft 45, into which the coaction pin 96 extends, is formed into such an elongated hole 97 as not to obstruct the rightward and leftward movements of the coaction pin 96.

The second control valve V_2 is composed of: a cylindrical follower valve spool 101, which is slidably fitted in the hollow follower pulley shaft 56; and a main valve spool 100 which is slidably fitted in that follower valve spool 101. The main valve spool 100 has its central portion formed with oil supply and release passages 103 and 104 which are isolated from each other by means of a partition 102. The oil supply passage 103 is made to communicate with the aforementioned second oil line L_2 formed in the cover C_3 by way of a communication conduit 105 inserted thereinto, whereas the oil release passage 104 is opened into the hollow portion of the follower pulley shaft 56, which has communication with the inside of the casing C.

The cover C_3 has its inner wall formed with an oil port 107 of the second oil line L_2, which is aligned with the axis of the follower pulley shaft 56. In that oil port 107, there is fitted by means of an O-ring 108 acting as an elastic seal member a mounting flange 106 which is welded to the outer circumference of the communication conduit 105 and which is prevented from coming out by means of a retainer ring 109. Thus, the communicating conduit 105 is supported in the cover C_3 in such a floating manner that it can follow the fluctuations of the axis of the follower pulley shaft 56 thereby to prevent the two members 56 and 105 from being worn due to their abutments.

On the other hand, the main valve spool 100 has its outer circumference formed with a pair of left and right annular oil supply grooves 110 and 111 and an annular oil release groove 112. Those oil supply grooves 110 and 111 are both made to communicate with the aforementioned oil supply passage 103 by way of holes 113 and 114, whereas the oil release groove 112 of the oil release system is made to communicate with the aforemention d oil rel ase passage 104 by way of a hole 115. On the other hand, the follower valve spool 101 has its outer circumference formed with a pair of left and right annular oil grooves 116 and 117. This right oil groove 117 is made to always communicate not only with the right oil supply groove 111 of the main valve spool 100 by way of a hole 118 but also with the first oil pressure chamber 62_1 of the hydraulic cylinder 62 by way of a hole 119, an annular oil passage 120 and an oil passage 121. The left oil groove 116 is made to always communicate with the second oil pressure chamber 62_2 of the hydraulic cylinder 62 by way of a hole 122. On the other hand, the follower valve spool 101 is formed with holes 123 and 124 for providing and blocking the communications between the left oil groove 116, and the left oil supply groove 110 and the oil release groove 112 of the main valve spool 100. Moreover, the follower valve spool 101 is connected to the movable pulley half 59 by means of a coaction pin 125, which extends radially of the follower pulley shaft 56, so that it can move to the right and left together with the latter. That portion of the follower pulley shaft 56, into which the coaction pin 125 extends, is so formed with an elongated hole 126 as not to obstruct the rightward and leftward movements of the coaction pin 56.

Both the first and second control valves V_1 and V_2 are so connected to the coaction mechanism 130 that the movable pulley halves 47 and 59 at the drive and driven sides may operate in synchronism with each other. That coaction mechanism 130 is constructed of: a push pin 131, which is interposed between and in parallel with the two control valves V_1 and V_2; a shifter 132 which is slidably borne on that push pin 131; and a coaction rod 133 which has its middle portion fixed to the shifter 132 and both its ends connected to the main valves spools 70 and 100 of the two control valves V_1 and V_2. The aforementioned shifter 132 is operated by the swinging motions of a shift lever 132 which is pivotally supported by the casing C. On the other hand, the shift lever 134 is operated by the turning operation of the left grip Hg of the steering handle H of Fig. 1.

Here, the operations of both the control valves V_1 and V_2 will be described in the following. In case the shifter 132 is positioned at its righthand limit in which it abuts against the cover C_3, as shown in Fig. 3, the first control valve V_1 has its hole 94 closed by the main valve spool 70 to block the communication between the right oil supply groove 82 and the right oil groove 87 but to provide the communication between the oil release groove 83 and the oil release port 95. Since the left oil supply groove 81 and the left oil groove 86 are always held in a state to communicate with each other, on the contrary, the working oil pressure waiting in the cylindrical oil passage 74 is introduced into the first oil pressure chamber 50_1 by way of the oil grooves 81, 86 and so on, whereas the second oil pressure chamber 50_2 is opened into the oil release port 95 by way of the oil grooves 82, 87 and so on. As a result, the piston 11 is moved to the right by the oil pressure of the first oil pressure chamber 50_1 until it holds the movable pulley half 47 at its rearmost limit.

In this case, moreover, the second control valve

V_2 has its left oil supply groov 110 communicating with the left oil groove 116 by way of the hole 123 and its hole 124 closed by the main valve spool thereby to block the communication between the oil release groove 112 and the left oil groove 116. Since the right oil supply groove 111 and the right oil groove 117 are always held in their communicating states, on the other hand, the working oil pressure waiting in the oil supply passage 103 is introduced into the first and second oil pressure chambers 62_1 and 62_2 of the hydraulic cylinder 62 so that the piston 61 is moved to the right by the oil pressure difference, as has been described hereinbefore, until it holds the movable pulley half 59 in its foremost limit.

Thus, since the effective radius of the drive V-pulley 40 is controlled to the minimum whereas the effective radius of the follower V-pulley 41 is controlled to the maximum, the former pulley 40 can drive the latter pulley 41 at the maximum reduction ratio.

Next, if the shift 132 is moved to the left, both the main valve spools 70 and 100 are simultaneously moved to the left by the action of the coaction rod 133. Moreover, when the hole 94 is opened by the leftward movement of the main valve spool 70 to provide the communication between the right oil supply groove 82 and the right oil groove 87 but to close the oil release port 95 by means of the main valve spool 70, then the working oil pressure of the cylindrical oil passage 74 is introduced into the second oil pressure chamber 50_2, too, so that the piston 49 starts to be moved to the left by the oil pressure difference, as has been described hereinbefore, thereby to move the movable pulley half 47 forward. Then, this advance of the movable pulley half 47 is transmitted through the coaction pin 96 to the follower valve spool 71 so that this valve spool 71 is simultaneously moved to follow the main valve spool 70. In acordance with this following operation, the hole 94 and the oil release port 95 are closed by the main valve spool 70 so that the piston 49 and, accordingly, the movable pulley half 47 is stopped when the second oil pressure chamber 50_2 is blocked from communicating with both the cylindrical oil passage 74 and the oil release port 95. In other words, the movable pulley half 47 can advance in accordance with the leftward movement of the main valve spool 70.

In accordance with this leftward movement of the main valve spool 100, on the other hand, the hole 123 is closed by the main valve spool 100 whereas the hole 124 is opened to provide the communication between the oil release groove 112 and the left oil groove 116 so that the oil pressure of the second oil pressure chamber 62_2 is released into the oil release passage 104. As a result, the piston 61 starts to be moved to the left by the oil pressure of the first oil pressure chamber 62_1 thereby to retract the movable pulley half 59. Then, the retraction of this movable pulley half 59 is transmitted through the coaction pin 125 to the follower spool valve 101 so that this valve 101 is simultaneously moved to accompany th main valve spool 100. By this accompanying operation, both the holes 113 and 114 are closed by the main valve spool 100 so that the second oil pressure chamber 62_2 is blocked from its communication with both the oil supply passage 103 and the oil release passage 104. Then, the piston 61 and, accordingly, the movable pulley half 59 are stopped. In other words, the movable pulley half 59 can retract in accordance with the leftward movement of the main valve spool 100.

Thus, since the advance of the movable pulley half 47 of the drive V-pulley 40 and the retraction of the movable pulley half 59 of the driven V-pulley 41 are conducted in synchronism with each other, the reduction of the effective radius of the drive V-pulley 42 and the enlargement of the effective area of the driven V-pulley 41 conducted in synchronism with each other without any application of an excessive tension to the V-belt 42 so that the reduction ratio between the two V-pulleys 40 and 41 can be reduced without fail thereby to elongate the lifetime of the V-belt 42 and to efficiently effect the power transmission. Moreover, since the main and follower valve spools 70 and 71, and 100 and 101 are made internally and externally dual and are disposed in the hollow drive and follower pulley shafts 45 and 56, they can be made compact. Furthermore, since the main and follower valve spools 70 and 71, and 100 and 101 are slidably fitted in the rotating hollow drive and driven pulley shafts 45 and 56, respectively, they are smooth in motions while eliminating no fear that they come to a standstill. Furthermore, since the first oil pressure chambers 50_1 and 62_1 are always held in communication with the oil pump P whereas only the second oil pressure chambers 50_2 and 62_2 have their communications switched to the oil pump P or the oil release system, the main and follow r valve spools 70 and 71, and 100 and 101 have their constructions simplified.

In the constructions thus far described, the hydraulic cylinder 50 of the drive V-pulley 40 is made to have a larger diameter than the hydraulic cylinder 62 of the driven V-pulley 41. According to these constructions, the hydraulic force to be received by the piston 49 at the drive side can always be made stronger than that to be received by the piston 61 at the driven side, even if the oil pressure is dropped, so that the responsiveness of the speed change can be effectively improved.

In the piston 49 of the drive V-pulley 40, moreover, the following inequality holds:

$$A_2 - A_1 > A_1,$$

if the pressure-receiving area at the side of the first oil pressure chamber 50_1 is designated at A_1 and if the pressure-receiving area at the side of the second oil pressure chamber 50_2 is designated at A_2.

In the piston 61 of th driven V-pulley 41, on the other hand, the following inequality holds:

$$B_2 - V_1 > B_1,$$

if the pressure-receiving ar a at the side of the first oil pressure chamber 62_1 is designated at B_1 and if the pressure-receiving area at the side of the second oil pressure chamber 62_2 is designated at B_2. As a result, the advancing forces of the respective movable pulley halves 47 and 59 by the oil pressures can always be made stronger than the retracting forces thereof. As a result, even if the forces for reducing the effective radii are applied to the respective movable pulley halves 47 and 59 by the centrifugal force acting upon the V-belt 42, the responsiveness to the speed changes in the directions to enlarge the effective radii of the movable pulley halves 47 and 59 can be improved.

In the hydraulic cylinders 50 and 62, moreover, there are disposed under compression the springs 52 and 67 which act as elastic members for urging the movable pulley halves 47 and 59 in their advancing directions, i.e., in the directions to have their effective radii enlarged. Those springs 53 and 67 function to apply a pre-tension to the V-belt 42 when no oil pressure is introduced yet into the respective hydraulic cylinders 50 and 62. As a result, it is possible to prevent any excessive shocking load from being exerted upon the V-belt 42 when in the starting operation thereby to elongate the lifetime of the V-belt 42. Moreover, since the elastic members, i.e., the springs 53 and 67 are accommodated in the hydraulic cylinders 50 and 62, no special space is required therefor so that the size of the continuous reduction gear mechanism can be effectively reduced.

Incidentally, if only one of the aforementioned springs 53 and 67 is provided, the pre-tension can be applied to the V-belt 42.

Since, in the auxiliary case C_2, the hydraulic actuator Hy1 of the drive V-pulley 40 is disposed in the outer front whereas the hydraulic actuator Hy2 of the driven V-pulley 41 is disposed at the inner back, the fixed pulley half 57 of the driven V-pulley 41 is necessarily disposed at the outer back of the same. Since the fixed pulley half 57 is equipped with no accessory such as the hydraulic cylinder 62, the casing C at the back of the fixed pulley half 57 is formed at its righthand outer rear portion with a recess 135 so that the contour of the casing C can be reduced. As shown in Fig. 1, a brake pedal Bp is disposed by making use of that recess 135. Thus, that brake pedal Bp can have its outward protrusion reduced or its protruding extent reduced. Incidentally, reference letter St appearing in the same Figure indicates a foot step.

On the other hand, the diagonal arrangements of the two hydraulic cylinders 50 and 62 with respect to the two V-pulleys 40 and 41 are advantageous because the external diameters of the respective hydraulic cylinders 50 and 62 can be freely set without any int rference with the other hydraulic cylinders 62 and 50 even in case they are disposed in the vicinity of the tw V-pulleys 40 and 41.

Next, the auxiliary reduction gear mechanism Ta will be described.

This auxiliary reduction gear mechanism Ta is constructed, as shown in Figs. 1 and 2, of: an input shaft 138 which is borne on the driven pulley shaft 56 between the bearings 64 and 65 by means of a needle bearing 137; and the output shaft 141 which has both its end portions borne in the main case C_1 at the back of the input shaft 138 by means of a needle bearing 139 and a ball bearing 140. The input shaft 138 is connected in one hand to the driven pulley shaft 56 through a reduction gear train 142 and on the other hand to the output shaft 141 through low- and high-speed gear trains 143 and 144.

The reduction gear train 142 is composed of: a first small gear 145 which is splined to the driven pulley shaft 56; a first large gear 147 which is driven by that small gear 145 through an intermediate gear 146; a second small gear 148 which is made rotatable together with that large gear 147; and a second large gear 149 which is driven by that small gear 148. The first large gear 147 and the second small gear 148 made integral with each other are borne on the output shaft 141 by means of a needle bearing 150, whereas the second large gear 149 is integrated with one end of the input shaft 138. As a result, the rotations of the driven pulley shaft 56 are decelerated one step by the first gears 145 and 147 and the other step by the second gears 148 and 149 until they can be transmitted to the input shaft 138.

The low-speed gear train 143 is composed of: a drive gear 151 which is made integral with the input shaft 138; and a driven gear 152 which is rotatably borne on the output shaft 141 such that it is driven by the aforementioned gear 151. On the other hand, the high-speed gear train 144 is constructed of: a drive gear 153 which is made integral with the input shaft 138; and a driven gear 154 which is rotatably borne on the output shaft 141 such that it is driven by the aforementioned gear 153. Moreover, the reduction ratio is naturally set at a higher level for the low-speed gear train 143 than for the high-speed gear train 144. To the output shaft 141, furthermore, there is slidably splined at 156 a shifter 155 which can be alternately coupled in a dog manner to the two driven gears 152 and 154. As a result, the shifter 155 has two shift positions, i.e., the slow position "Lo", in which it is coupled to the driven gear 152, and the fast position "Hi", in which it is coupled to the driven gear 154, and can additionally have a neutral position "N" in which it is coupled to neither the driven gears 152 and 154. This shifting operations of the shifter 155 are effected by a shift fork 157. Thus, if the shifter 155 is shifted to the position "Lo" or "Hi", either the low-speed gear train 143 or the high-speed gear train 144 is brought into its operative state so that reduction ratios of two high and low steps can be effected betwe n the input and output shafts 138 and 141.

Th auxiliary reducti n gear mechanism Ta thus far described c mpensates the short r duc-

tion ratio range of the aforementioned continuous reduction gear mechanism Tm, which is caused by the reduction in the external diameter of the drive V-pulley 40. In other words, by the addition of the auxiliary reduction gear mechanism Ta, the drive and driven V-pulleys 40 and 41 can have their axial distance shortened as much as possible and their sizes reduced so that the power unit Pu can be accommodated compactly in the casing C while allowing the accompanying more or less sacrifice of the reduction ratio range of the continuous reduction gear mechanism Tm.

On the other hand, the external diameter of the start clutch Sc is determined by the number, the internal and external diameters and the surface pressures of the drive and driven friction discs 6 and 7. For example, the external diameter of the start clutch Sc can be reduced by increasing the number of those friction discs 6 and 7 so that their surface pressures may be augmented. With this construction, however, there arises a disadvantage that the operations are not smooth or that the operating load is augmented.

In the present invention, since the external diameter of the start clutch Sc is reduced to substantially the same as that of the drive V-pulley 40 and since the start clutch Sc is constructed of a hydraulic multi-disc clutch, the start clutch Sc has its size reduced and its operation smoothened, and its operating load is reduced.

Reference is directed to co-pending applications numbers 82306278 and 85114299 in which a belt type continuous reduction mechanism as described above is also described.

Claims

1. A belt type continuous reduction gear mechanism comprising a drive V-pulley (40) arranged coaxially with a start clutch (Sc) which is arranged coaxially with the crankshaft (1) of an engine and is constructed as a disc clutch; a driven V-pulley (41) connected to an auxiliary gear mechanism (Ta) of plural-step type; a V-belt (42) made to run under tension between the two V-pulleys (40, 41); a casing (C) accommodating at least the engine (E), the start clutch and both drive and driven V-pulleys (40, 41) each of which is equipped with a movable pulley half (47, 59) that can move back and forth relative to a fixed pulley half (44, 57); a drive pulley shaft (45) arranged coaxially with the crankshaft (1) and on which are mounted both the fixed pulley half (44) of the drive V-pulley (40) at the inner side and the movable pulley half (47) of the drive V-pulley (40) at the outer side, this movable pulley half (47) carrying an actuator (H_{y1}) for adjusting the effective radius and this fixed pulley half (44) being connected to the start clutch; and a driven pulley shaft (56) arranged in parallel with the drive pulley shaft (45) and on which are mounted both the fixed pulley half (57) of the driven V-pulley (41) at the outer side and the movable pulley half (59) of the driven V-pulley (41) at the inner side, this movable pulley half (59) carrying an actuator (H_{y2}) for adjusting the effective radius, characterised in that the drive V-pulley (40) has its external diameter r duced to substantially the same as that of the start clutch thereby to set the reduction ratio range at a relatively small value; and in that the auxiliary gear mechanism (Ta) is of plural step reduction type for compensating the shortage of said reduction ratio range; wherein the start clutch is of hydraulic multi-disc type; wherein an oil pump (P) is connected through a control line (Lc) to the start clutch; and wherein there is disposed on the control line (Lc) a clutch valve (Vc) for controlling the discharge pressure of the oil pump (P) so that the connection oil pressure of the start clutch is varied proportionately of the squeezed extent of a clutch lever (32).

2. A belt type continuous reduction gear mechanism (Tm) as claimed in claim 1, wherein the start clutch comprises an outer clutch member (4) connected to an end portion of the crankshaft (1) to rotate therewith; an inner clutch member (5) fast with the fixed pulley half (44) of the drive V-pulley (40); a plurality of drive and driven friction discs (6, 7) alternately interposed in a coextensive manner between the outer clutch member (4) and the inner clutch member (5) and slidably coupled to the outer and inner clutch members (4, 5), respectively; a hydraulic actuator mechanism (9, 11) disposed in the outer clutch member (4) for thrusting the drive and driven friction discs (6, 7); a working oil passage (14) formed in said end portion of the crankshaft (1) for supplying the working oil to the hydraulic actuator mechanism (9, 11); a cooling oil passage (19, 3, 20) branched from the working oil passage (14) for supplying the working oil as cooling oil to the drive and driven friction discs (6, 7); and a flow regulating valve (15) slidable fitted in the working oil passage (14) and made operative in response to the oil pressure in this oil passage (14) to reduce the flow rate of the cooling oil passage (19, 3, 20) to a zero or low level when the start clutch is blocked or connected, or to increase the flow rate of the cooling oil passage (19, 3, 20) to a high level when the start clutch is partially applied.

3. A belt type continuous reduction gear mechanism (Tm) as claimed in claim 2, wherein said end portion of the crankshaft (1) is hollow and is borne in a casing (C) having an oil supply passage, and wherein a communication conduit (72) for supplying the working oil to the hydraulic actuator mechanism (9, 11) has one end supported in a floating manner through an elastic seal member (79) in an oil port of the oil supply passage, which is formed in the inner wall of the casing (C) and which has its centre line aligned with the axis of said end portion of the crankshaft (1), and its other end fitted in said end portion of the crankshaft (1).

Pat ntansprüche

1. Stufenloses Riemengetriebe mit einer trei-

benden Keilriemenscheibe (40), die koaxial mit einer Startkupplung (Sc) angeordnet ist, die ihrerseits koaxial mit der Kurbelwelle (1) eines Motors angeordnet ist und als eine Scheibenkupplung ausgeführt ist, einer angetriebenen Keilriemenscheibe (41), die mit einem Hilfsgetriebemechanismus (Ta) des Mehrstufentyps verbunden ist, einem Keilriemen (42), der dazu bestimmt ist, unter Spannung zwischen den beiden Keilriemenscheiben (40, 41) zu laufen, einem Gehäuse (C), das zumindest aufnimmt: den Motor (E), die Startkupplung und sowohl die antreibende als auch die angetriebene Keilriemenscheibe (40, 41), wovon jede mit einer bewegbaren Riemenscheibenhälfte (47, 59) versehen ist, die sich rückwärts und vorwärts relativ zu einer feststehenden Riembenscheibenhälfte (44, 57) bewegen kann, einer antreibenden Riemenscheibenwelle (45), di koaxial mit der Kurbelwelle (1) angeordnet ist und auf welcher sowohl die festehende Riemenscheibenhälfte (44) der antreibenden Keilriemenscheibe (40) auf der inneren Seite als auch die bewegbare Riemenscheibenhälfte (47) der antreibenden Keilriemenscheibe (40) auf der äußeren Seite montiert sind, wobei diese bewegbare Riemenscheibenhälfte (47) einen Versteller (H_{y1}) zum Justieren des effektiven Radius trägt und diese feststehende Riemenscheibenhälfte (44) mit der Startkupplung verbunden ist, und einer angetriebenen Riemenscheibenwelle (56), die parallel zu der antreibenden Riemenscheibenwelle (45) angeordnet ist und auf welcher sowohl die feststehende Riemenscheibenhälfte (57) der angetriebenen Keilriemenscheibe (41) auf der äußeren Seite als auch die bewegbare Riemenscheibenhälfte (59) der angetriebenen Keilriemenscheibe (41) auf der inneren Seite montiert sind, wobei diese bewegbare Riemenscheibenhälfte (59) einen Versteller (H_{y2}) zum Justieren des effektiven Radius trägt, dadurch gekennzeichnet, daß der äußere Durchmesser der antreibenden Keilriemenscheibe (40) auf im wesentlichen denselben wie dem der Startkupplung reduziert ist, um dadurch den Reduktionsverhältnisbereich bei einem relativ kleinen Wert einzustellen, und daß der Hilfsgetriebemechanismus (Ta) vom Mehrstufenreduktionstyp zum Kompensieren der Knappheit des Reduktionsverhältnisbereichs ist, wobei die Startkupplung vom Hydraulik-Merhrscheibentyp ist, wobei eine Ölpumpe (P) durch eine Steuerleitung (Lc) mit der Startkupplung verbunden ist und wobei in der Steuerleitung (Lc) ein Kupplungsventil (Vc) zum Regeln des Austrittsdrucks der Ölpumpe (P) angeordnet ist, so daß der Verbindungsöldruck der Startkupplung proportional zu dem Ausmaß des Drucks auf einen Kupplungshebel (32) variiert wird.

2. Stufenloses Riemengetriebe (Tm) nach Anspruch 1, bei dem die Startkupplung umfaßt: ein äußeres Kupplungsteil (4), das mit einem Endabschnitt der Kurbelwelle (1) verbunden ist, um sich mit dieser zu drehen, ein inneres Kupplungsteil (5), das an der feststehenden Riemenscheibenhälfte (44) der antreibenden Keilreim nscheibe (40) befestigt ist, eine Vielzahl von antreibenden und angetriebenen Reibungsscheiben (6, 7), di abwechselnd mit gleichem Umfang zwischen dem äußeren Kupplungsteil (4) und dem inneren Kupplungsteil (5) angeordnet und verschiebbar mit dem äußeren bzw. dem inneren Kupplungsteil (4, 5) verbunden sind, einem hydraulischen Betätigungsmechanismus (9, 11), der in dem äußeren Kupplungsteil (4) zum Zusammendrücken der treibenden und der angetriebenen Reibungsscheiben (6, 7) angeordnet ist, einem Arbeitsölkanal (14), der in dem Endabschnitt der Kurbelwelle (1) zum Liefern des Arbeitsöls zu dem hydraulischen Betätigungsmechanismus (9, 11) ausgebildet ist, einem Kühlölkanal (19, 3, 20), der von dem Arbeitsölkanal (14) zum Liefern des Arbeitsöls als Kühlöl an die treibenden und angetriebenen Reibungsscheiben (6, 7) abgezweigt ist, und einem Strömungsregulierungsventil (15), das verschiebbar in den Arbeitsölkanal (14) eingepaßt ist und in Reaktion auf den Öldruck in diesem Ölkanal (14) wirksam ist, um die Strömungsrate des Kühlölkanals (19, 3, 20) bis auf einen Null- oder Niedrigwert zu reduzieren, wenn die Startkupplung blockiert oder durchgekuppelt ist, oder um die Strömungsrate des Kühlölkanals (19, 3, 20) bis auf einen hohen Wert zu erhöhen, wenn die Startkupplung teilweise wirksam ist.

3. Stufenloses Reimengetriebe (Tm) nach Anspruch 2, bei dem der Endabschnitt der Kurbelwelle (1) hohl ist und in einem Gehäuse (C) gelagert ist, das einen Ölzuführungskanal hat, und bei dem ein Verbindungskanal (72) zum Zuführen des Arbeitsöls zu dem hydraulischen Betätigungsmechanismus (9, 11) ein Ende aufweist, das in einer schwimmenden Weise durch ein elastische Dichtungselement (9) in einem Öleinlaß des Ölzuführungskanals gehalten ist, der in der inneren Wandung des Gehäuses (C) ausgebildet ist und seine Mittellinie mit der Achse des Endabschnitts der Kurbelwelle (1) ausgerichtet hat, und dessen anderes Ende in den Endabschnitt der Kurbelwelle (1) eingepaßt ist.

Revendications

1. Variateur de vitesse à courroie comprenant une poulie menante en V (40) monté de manière coaxiale avec un mècanisme d'embrayage pour le démarrage (Sc) qui est monté de manière coaxiale avec le vilebrequin (1) d'un moteur et se présente comme un embrayage à disque; une poulie menée en V (41) reliée à un mécanisme d'engrenage auxiliaire (Ta) du type à plusieurs pas; une courroie en V (42) conçue pour être entraînée en condition tendue entre les deux poulies en V (40, 41); un carter (C) logeant au moins le moteur (E), le mécanisme d'embrayage pour le démarrage et les poulies en V menée et menante (40, 41), chacune d'entre elles étant pourvue d'une demi-poulie mobile (47, 59) pouvant être déplacée vers l'avant et vers l'arrière par rapport à une demi-poulie fixe (44, 57); un arbre de poulie menante (45) monté de manière coaxiale avec le vilebrequin (1) et sur lequel sont

montées à la fois la demi-poulie fixe (44) de la poulie menante en V (40) du côté intérieur et la demi-poulie mobile (47) de la poulie menante en V (40) du côté extérieur, cette demi-poulie mobile (47) portant un actionneur (H_{y1}) pour régler le rayon effectif et cette demi-poulie fixe (44) étant reliée à l'embrayage; et un arbre de poulie menée (56) monté en parallèle avec l'arbre de poulie menante (45) et sur lequel sont montées à la fois la demi-poulie fixe (57) de la poulie menée en V (41) du côté extérieur et la demi-poulie mobile (59) de la poulie menée en V (41) du côté intérieur cette demi-poulie mobile (59) portant un actionneur (H_{y2}) pour régler le rayon effectif; caratérisé en ce que le diamètre extérieur de la poulie menante en V (40) est réduit pratiquement à celui de l'embrayage pour régler la plage de la réduction de la vitesse à une valeur relativement faible; et en ce que ledit mécanisme d'engrenage auxiliaire (Ta) est du type à renvoi à plusieurs pas pour compenser la diminution de ladite plage de la réduction de vitesse; dans lequel le mécanisme d'embrayage pour le démarrage est du type hydraulique multidisques; dans lequel une pompe à huile (P) est reliée par une ligne de commande (Lc) à l'embrayage; et dans lequel une soupape d'embrayage (Vc) est placée sur la ligne de commande (Lc) pour commander la pression de sortie de la pompe à huile (P) de telle sorte que la pression d'huile de liaison de l'embrayage est modifiée proportionnellement à la pression exercée sur une pédale de débrayage (32).

2. Variateur de vitesse à courroie (Tm) selon la revendication 1, dans lequel le mécanisme d'embrayage pour le démarrage comprend un organe d'embrayage extérieur (4) relié à une partie d'extrémité d'un vilebrequin (1) pour tourner avec celui-ci; un organe d'embrayage intérieur (5) attaché à la demi-poulie fixe (44) de la poulie menante en V (40); une pluralité de disques de friction (6, 7) menés et menants intercalés alternativement de manière à être étendus ensemble entre l'organe d'embrayage extérieur (4) et l'organe d'embrayage intérieur (5) et couplés de manière coulissante aux organes d'embrayage extérieur et intérieur (4, 5) respectivement; un mécanisme actionneur hydraulique (9, 11) placé dans l'organe d'embrayage extérieur (4) pour exercer une poussée sur les disques de friction mené et menant (6, 7); un conduit d'huile de travail (14) ménagé dans ladite partie d'extrémité du vilebrequin (1) pour alimenter le mécanisme actionneur hydraulique (9, 11) en huile de travail; un conduit d'huile de refroidissement (19, 3, 20) partant du conduit d'huile de travail (14) pour amener l'huile de travail servant d'huile de refroidissement aux disques de friction menant et mené (6, 7); et une soupape de régulation d'écoulement (15) placée de manière coulissante dans le conduit d'huile de travail (14), dont le fonctionnement dépend de la pression d'huile dans ce conduit d'huile (14), pour amener la vitesse d'écoulement dans le conduit d'huile de refroidissement (19, 3, 20) à un niveau faible ou arr^ter l'écoulement lorsque l'embrayage est arrêté ou embrayé, ou amener la vitesse d'écoulement dans le conduit d'huile de refroidissement (19, 3, 20) à un niveau élevé lorsque l'embrayage est applique partiellement.

3. Variateur de vitesse à courroie (Tm) selon la revendication 2, dans lequel ladite partie d'extrémité du vilebrequin (1) est creuse et est portée dans un carter (C) pourvu d'une conduite d'alimentation d'huile, et dans lequel une extrémité d'un conduit de communication (72) pour alimenter le mécanisme actionneur hydraulique (9, 11) en huile de travail est supportée de manière flottante à travers un organe d'étanchéité élastique (79) dans un orifice du conduit d'alimentation d'huile qui est ménagé dans la paroi intérieure du carter (C) et dont la ligne médiane est alignée avec l'axe de ladite partie d'extrémité du vilebrequin (1), son autre extrémité étant logée dans ladite partie d'extrémité du vilebrequin (1).

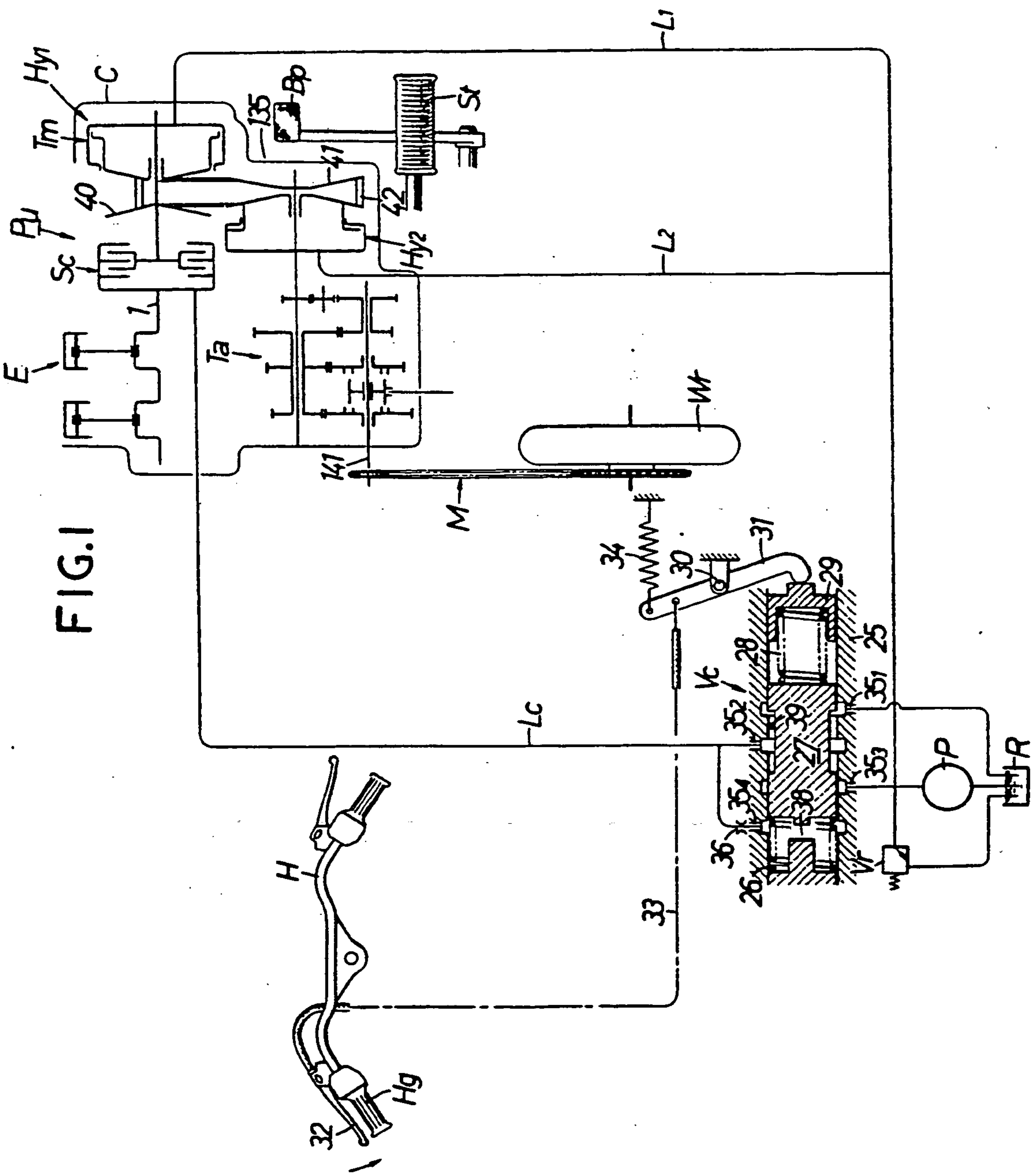
Hy1
C
Tm
40
Pu
Sc
E
1
Ta
141
M
Wr
L1
Bp
St
135
41
42
Hy2
L2
Lc
34
30
31
29
25
28
Vc
35_1
35_2
39
27
35_3
P
R
35_4
38
36
26
Vr
H
33
Hg
32

FIG.1

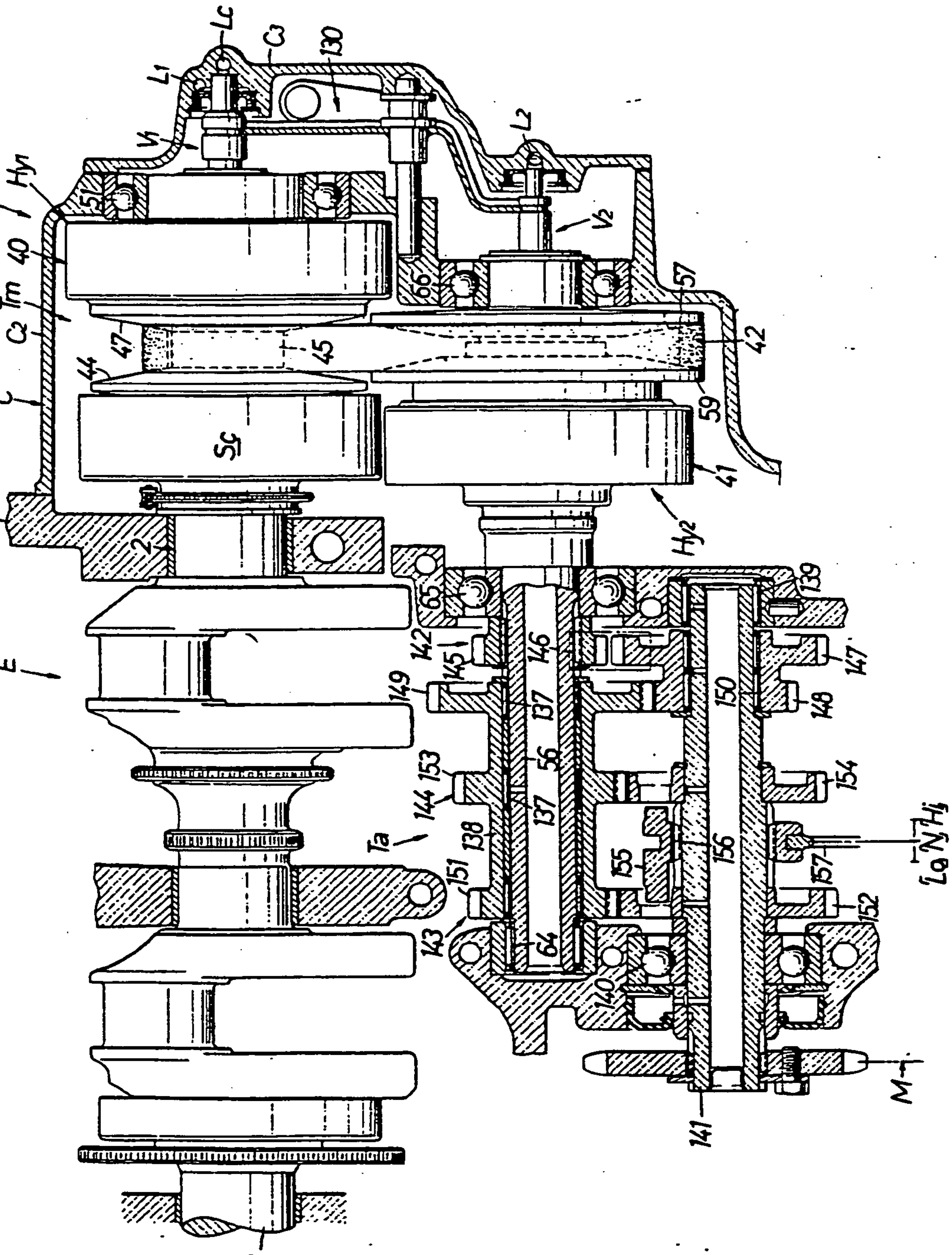
E
Lc
C3
130
L1
V1
L2
Hy1
Pu
51
V2
40
Tm
66
C2
57
47
45
42
44
C
59
Sc
41
C1
2
Hy2
65
139
142
145
146
147
149
137
150
148
56
153
154
144
Ta
138
155
156
La N Hi
151
157
152
143
64
140
M
141
1

FIG.2

FIG.3